

Purdue University
Purdue e-Pubs

International Compressor Engineering Conference

School of Mechanical Engineering

1976

Power Requirements and Associated Effects of Reciprocating Compressor Cylinder Ends Deactivated by Internal Bypassing

W. Hartwick

Follow this and additional works at: <https://docs.lib.purdue.edu/icec>

Hartwick, W., "Power Requirements and Associated Effects of Reciprocating Compressor Cylinder Ends Deactivated by Internal Bypassing" (1976). *International Compressor Engineering Conference*. Paper 184.
<https://docs.lib.purdue.edu/icec/184>

This document has been made available through Purdue e-Pubs, a service of the Purdue University Libraries. Please contact epubs@purdue.edu for additional information.

Complete proceedings may be acquired in print and on CD-ROM directly from the Ray W. Herrick Laboratories at <https://engineering.purdue.edu/Herrick/Events/orderlit.html>

POWER REQUIREMENTS AND ASSOCIATED EFFECTS OF RECIPROCATING¹.
COMPRESSOR CYLINDER ENDS DEACTIVATED BY INTERNAL BYPASSING

W. HARTWICK, Chief Designer - Cylinders
Cooper Energy Services
A Division Of Cooper Industries, Inc.
Mount Vernon, Ohio

INTRODUCTION

A common method of load/capacity control for heavy, continuous-duty reciprocating compressors where large steps are required is cylinder end deactivation accomplished by internal bypassing wherein the compression space is communicated with suction or discharge gas passage as in Fig. 1(a). The resulting cycle, Fig. 1(b),² reduces capacity of the deactivated end to zero. However, pressure drops, ΔP , incurred in pulling the gas into and pushing it out of the compression space through the bypass port(s) result in a definite power input requirement.

Recognition of this power requirement and associated effects are necessary for accurate performance prediction.

This paper will develop a general mathematical expression, which, when provided with appropriate experimentally determined bypass resistance factors, will permit calculation of the power loss incurred for each of several mechanical configurations of bypass deactivation. The effect of deactivated ends on capacity of active ends operating in parallel with them will be evaluated. Other side effects and considerations pertinent to bypass deactivation will also be covered

MECHANICAL CONFIGURATIONS

Common mechanical configurations to accomplish cylinder end deactivation are illustrated in Figs. 2 through 7. In those cases where the device may be actuated during operation, half of the figure shows the bypass mode and the other half illustrates the compression mode. Actuation is not necessarily limited to method illustrated.

Table 1 summarizes pertinent features.

THE COMMUNICATING BYPASS PORT AS AN ORIFICE

Fundamental to calculation of the instantaneous pressure drops throughout the cycle in Fig. 1(b) is knowledge of the pressure drop characteristic or resistance factor, R_F , of the port for flow in both the "in" and "out" directions. Resistance factors may be determined experimentally by establishing a steady flow of gas, usually air for convenience, through the bypass and measuring flow, relevant pressure drops, pressures, and temperatures as indicated in Fig. 8. A compressor cylinder with piston removed is used as a test fixture. By installing identical bypasses on the inlet and outlet sides as illustrated, data for "in" and "out" resistance factors may be obtained simultaneously.

1. The paper was originally presented at the American Society of Mechanical Engineers' Diesel and Gas Engine Power Conference and Exhibit, New Orleans, Louisiana, April 6 - 10, 1975.

2. Referring to Fig. 1(b), the power requirement derivation included in this paper assumes the existence of a constant pressure, represented by the dashed line, in the gas passage into which bypassing is directed. Departures from this assumption are discussed in the text.

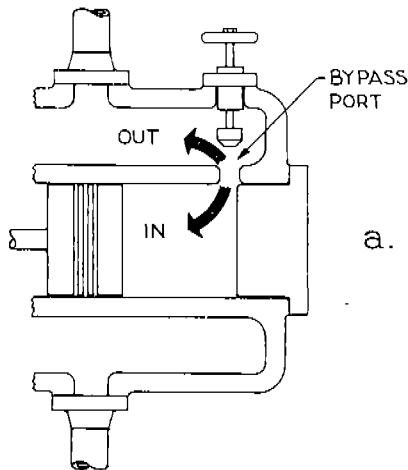


Fig. 1(a) Schematic of reciprocating compressor cylinder equipped with bypass device for deactivation

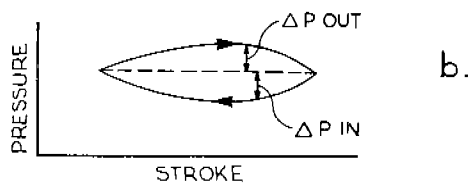


Fig. 1(b) Cycle in compression space of deactivated cylinder end. Arrows indicate direction of piston motion as related to Fig. 1(a). Dashed line is passage pressure

By definition

$$R_f = \frac{\Delta P_{obs.}}{\Delta P_{ideal}} \quad (1)^3.$$

$\Delta P_{obs.}$ is determined experimentally by means of the steady flow tests described in the foregoing.

ΔP_{ideal} is developed from head meter principles. Neglecting the velocity of approach factor and assuming unit discharge coefficient,

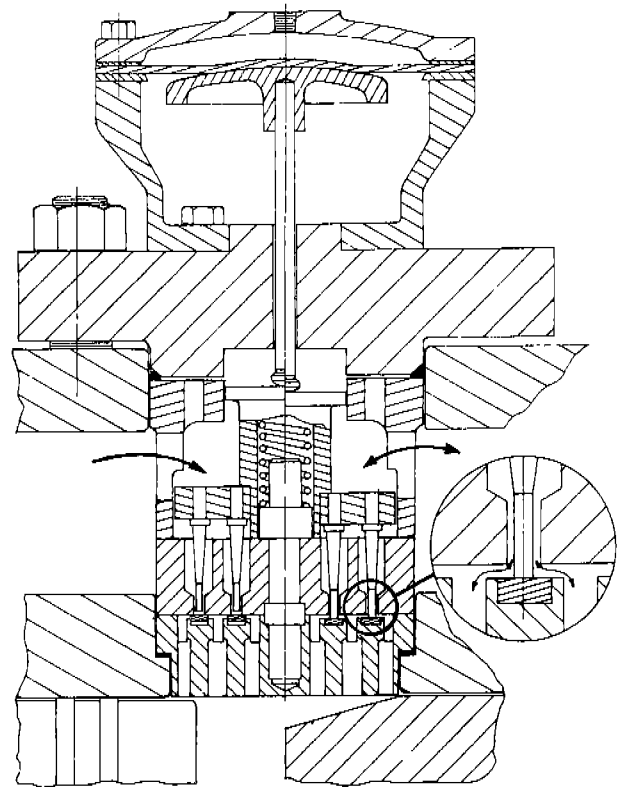


Fig. 2 Finger-type suction valve unloader. Valve plate is held off seat of conventional valve assembly during entire cycle to accomplish deactivation

$$Q = A_{port} \sqrt{2gh} \times 10^{-6} \quad (2)$$

$$(Q = \frac{60}{144} A_{port} \sqrt{2gh} \text{ in customary U.S. units.})$$

Rearranging,

$$h = \frac{5 \times 10^{11} Q^2}{g A_{port}^2} \quad (3)$$

(Constant = 2.88 for customary U.S. units)

3. All equations and quantities are expressed in Standard International (S.I.) units. Where applicable, corresponding equations and/or constants are shown parenthetically in customary U.S. units.

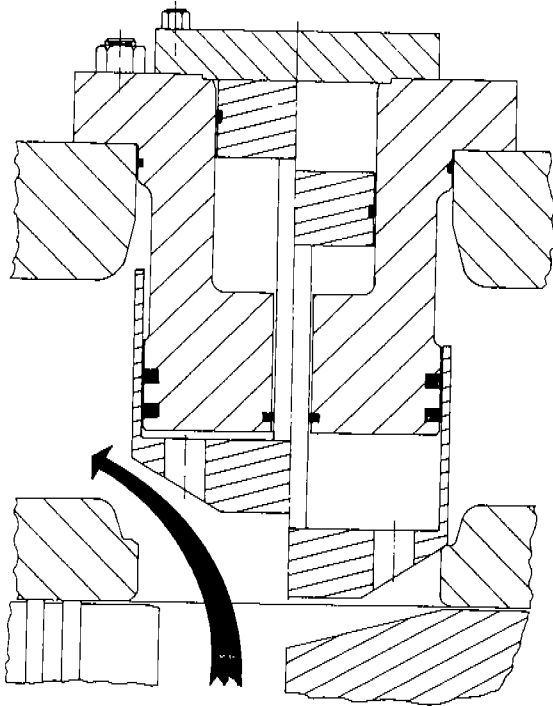


Fig. 5 Valve pocket bypass. Bypass occupies entire valve pocket; no conventional valving provided. Plug is lifted for deactivated mode; seated for compression mode. Applicable only if cylinder end has more than one suction or discharge valve pocket

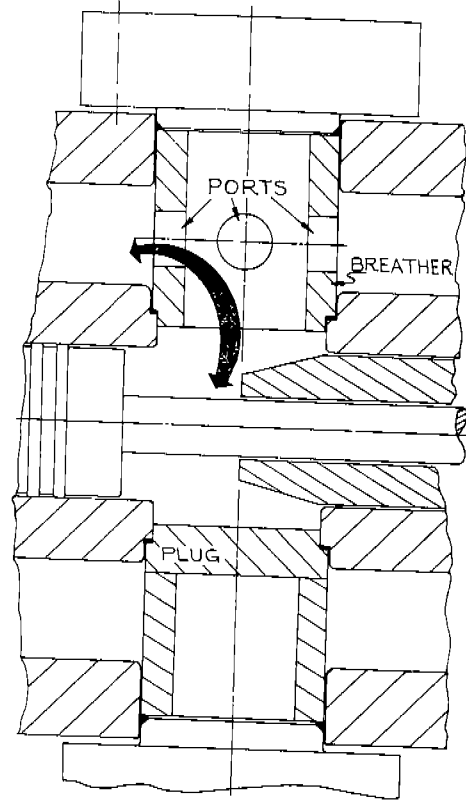


Fig. 6 Breather-plug bypass system. No conventional valving provided. Pockets on one side (suction or discharge) are open to gas passage; other side is provided with plug

Fig. 9 shows results of laboratory determination of resistance factors for the plug-type valve unloader (Fig. 3) for flow in both the in and out directions. Lift was varied to provide guidance for the designer. On the basis of this work, a 1.0 ratio of lift area to port was selected as an optimum compromise between resistance factor and the physical limitations related to plug lift.

PREDICTING POWER REQUIREMENT FOR DYNAMIC OPERATION

Under dynamic compressor operating conditions,

$$Q = V_{\text{pist}} \cdot A_{\text{pist}} \times 10^{-6} \quad (10)$$

(constant = 6.94×10^{-3})

Making provision for n bypass ports and substituting $\Delta P_{\text{pred.}}$ for $\Delta P_{\text{obs.}}$, the following is obtained from equations (9) and (10):

$$\Delta P_{\text{pred.}} = \frac{6.01 \times 10^{-5} R_f PM}{ZT} \times \left(\frac{V_{\text{pist.}} \cdot A_{\text{pist.}}}{n \cdot A_{\text{port}}} \right)^2 \quad (11)$$

(constant = 2.79×10^{-9})

By definition, $\Delta P_{\text{pred.}}$ is the instantaneous pressure drop corresponding to $V_{\text{pist.}}$ at a particular point in the stroke. The mean pressure drop over the entire stroke (in or out) is more useful and is given by the following variation of equation (11):

$$\Delta P_{\text{pred. mean}} = \frac{6.01 \times 10^{-5} R_f PM}{ZT} \left(\frac{A_{\text{pist.}}}{n \cdot A_{\text{port}}} \right)^2 \times \int_{S=0}^{S=1} \frac{V_{\text{pist.}}^2 \cdot dS}{S} \quad (12)$$

(constant = 2.79×10^{-9})

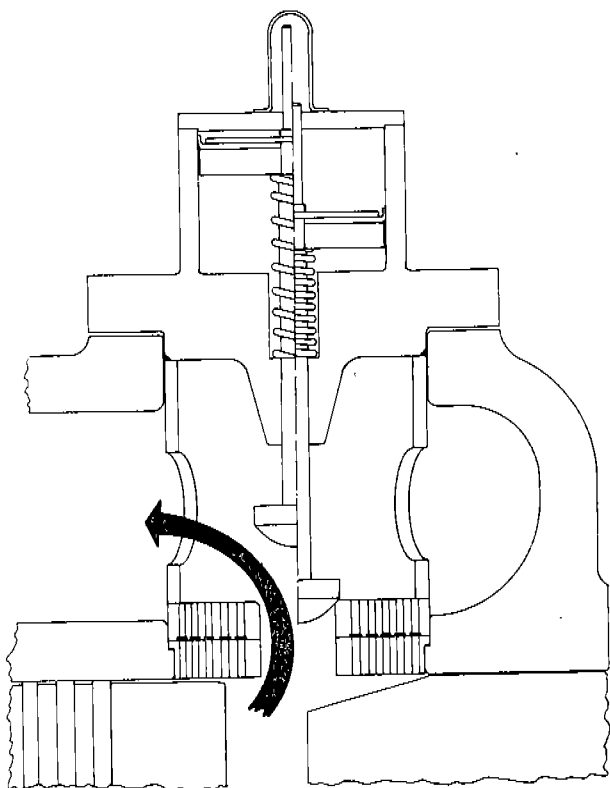


Fig. 3 Plug-type valve unloader. Conventional valve assembly modified to incorporate bypass port. Plug is lifted for deactivated mode; closed for compression mode

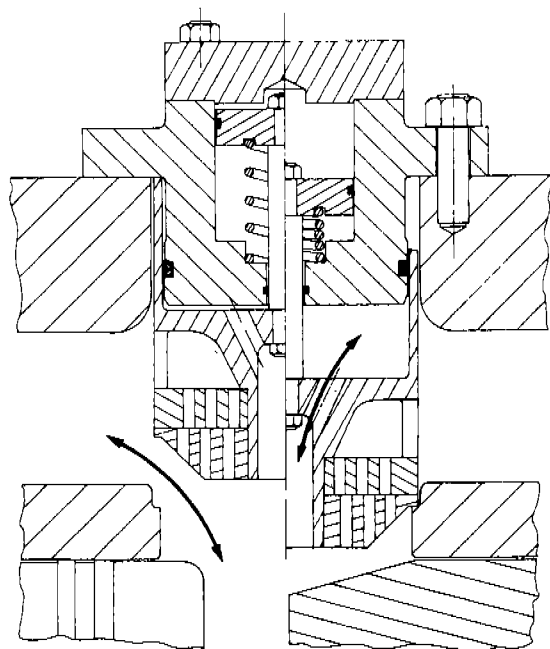


Fig. 4 Liftable valve assembly. Valve is modified to accomplish partial force balancing. Entire assembly is lifted for deactivated mode; seated for compression mode

Multiplying both sides of equation (3) by ρ_g (by ρ in customary U.S. units),

$$\rho_{gh} = \frac{5 \times 10^{11} \rho_Q^2}{A_{\text{port}}^2} \quad (4)$$

$$\left(\rho_h = \frac{2.88 \rho_Q^2}{g A_{\text{port}}^2} \right)$$

But,

$$\begin{aligned} \rho_{gh} &= \Delta P_{\text{ideal}} \times 10^3 \\ \left(\rho_h &= 144 \Delta P_{\text{ideal}} \right) \end{aligned} \quad (5)$$

Thus, combining equations (4) and (5),

$$\begin{aligned} \Delta P_{\text{ideal}} &= \frac{5 \times 10^8 \rho_Q^2}{A_{\text{port}}^2} \\ (\text{constant} &= 6.21 \times 10^{-4}) \end{aligned} \quad (6)$$

Also, from the equation of state for gases,

$$\rho = \frac{PM}{RT} \quad (7)$$

(multiplying constant = 144 for customary U.S. units).

From equations (6) and (7)

$$\begin{aligned} \Delta P_{\text{ideal}} &= \frac{6.01 \times 10^7 PM_Q^2}{RT A_{\text{port}}^2} \\ (\text{constant} &= 5.79 \times 10^{-5}) \end{aligned} \quad (8)$$

Combining equations (1) and (8), R_f can be expressed in terms of known and experimental data.

$$\begin{aligned} R_f &= \frac{1.66 \times 10^{-8} \Delta P_{\text{obs}} RT A_{\text{port}}^2}{PM_Q^2} \\ (\text{constant} &= 1.73 \times 10^4) \end{aligned} \quad (9)$$

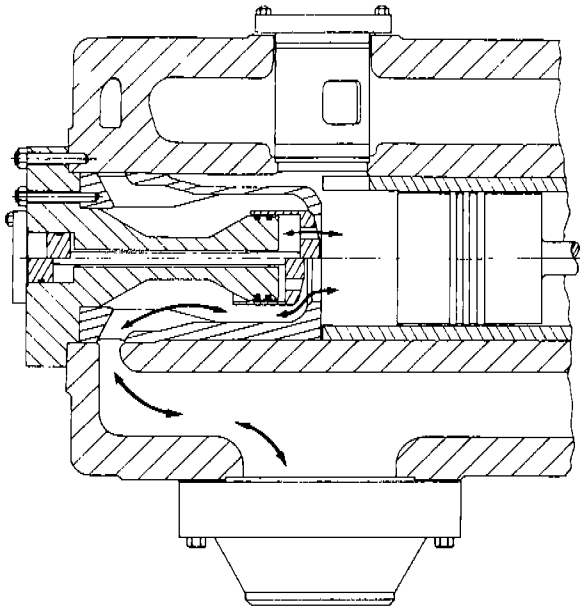


Fig. 7 Head end bypass. Cylinder is modified to accommodate large area, force-balanced plug. Plug is open for deactivated mode; closed for compression mode

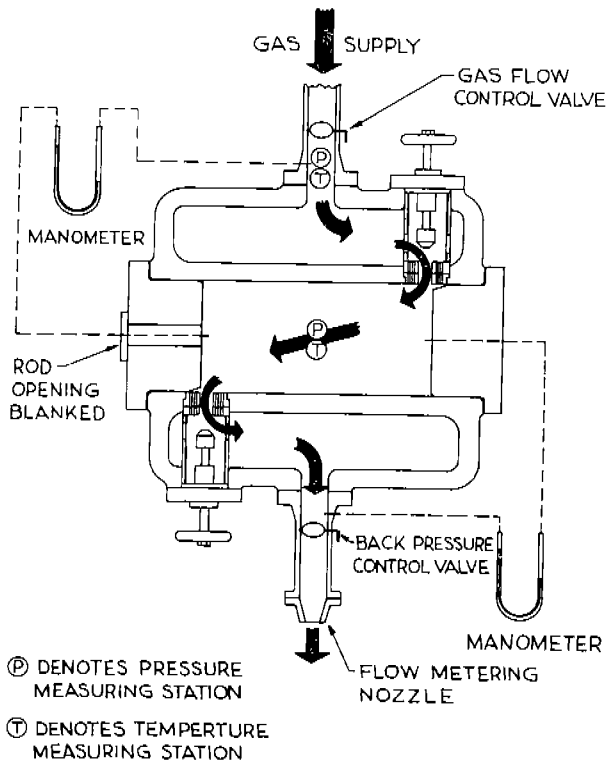


Fig. 8 Schematic for experimental determination of bypass resistance factor

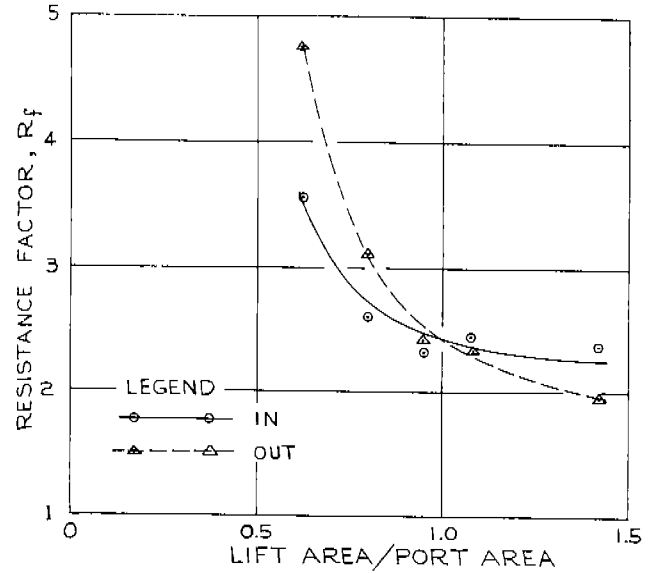


Fig. 9 Results of laboratory determination of resistance factor for plug-type valve unloader illustrated in Fig. 3

For a typical connecting rod length to crank radius of 5.0,

$$\int_{S=0}^{S=1} \frac{V_{\text{pist.}}^2}{S} dS = 1.87 \times 10^{-9} (S \cdot \text{rpm})^2 \quad (13)$$

(constant = 4.68×10^{-2})

Combining equations (12) and (13),

$$\Delta P_{\text{pred. mean}} = \frac{1.12 \times 10^{-13} R_f \text{PM}}{ZT} \times \left(\frac{A_{\text{pist.}}}{n \cdot A_{\text{port}}} \right)^2 (S \cdot \text{rpm})^2 \quad (14)$$

(constant = 1.31×10^{-10})

In a compressor, the work requirement for the in or out stroke is:

$$L = \Delta P_{\text{pred. mean}} A_{\text{pist.}} \cdot S \times 10^{-6} \quad (15)$$

(constant = 8.33×10^{-2})

The corresponding power requirement for the in or out stroke is:

$$U = 1.67 \times 10^{-5} L \cdot \text{rpm} \quad (16)$$

(constant = 3.03×10^{-5})

Combining equations (14), (15), and (16) and modifying to cover the total cycle power requirement for the cylinder end ("in" plus "out" stroke):

Table 1 Features of Bypass Deactivation Methods

Illustrated By Fig. No.	2	3	4	5	6	Not Illustrated	7
Description	Finger-Type Suction Valve Unloader	Plug-Type Valve Unloader	Liftable Valve Assembly	Valve Pocket Bypass	Breacher-Plug Bypass System	Removal Of Valve Plates And Springs	Head End Bypass
Applicable To	Suction Only	Suction Or Discharge	Suction Or Discharge	Suction Or Discharge	Suction Or Discharge	Suction Or Discharge	Suction Or Discharge
Applied To Valve Pocket?	Yes	Yes	Yes	Yes	Yes	Yes	No
Combined With Active Valve Area?	Yes	Yes	Yes	No	No	No	--
Capable Of Going From Compression Mode To Deactivated Mode And Vice Versa During Operation?	Yes	Yes	Yes	Yes	No	No	Yes
Full Complement Recommended?	Yes	Yes	Not Necessarily	No	Not Necessarily	Yes	--
Advantages	Retains Full Active Valve Area	Avoids Mechanical Control Of Valve Plate Action	Low Power Requirement In Deactivated Mode	Low Power Requirement In Deactivated Mode	Low Power Requirement	No Special Hardware Required	Low Power Requirement In Deactivated Mode; Independent Of Valves
Disadvantages	High Power Requirement In Deactivated Mode	High Power Requirement In Deactivated Mode; Reduced Active Valve Area; Adds Fixed Clearance	Reduced Active Valve Area; Adds Fixed Clearance	No Active Valve Area; Adds Fixed Clearance	Requires Shut-down and Mechanical Modification To Activate	Requires Shut-down and Mechanical Modification To Activate	Not Applicable To Crank End Adds Fixed Clearance
Principal Applicability	Low Operating Time In Deactivated Mode	Low Molecular Weight Service	High Operating Time In Deactivated Mode	Use Only In Generously Valved Cylinders ^a	Permanent Or Long Term Deactivation	Temporary Emergency Deactivation	Where Extensive And Finely Tuned Unloading Is Required
Definition Of A _{port}	Lift Area Of Valve	Area Of Bypass Hole Through Valve	Valve Pocket Area	Valve Pocket Area	Area Of Ports In Breather ^b	See Footnote d.	See Fig. 7
Resistance Factor, R _{fin} + R _{fout}	Lift Inches R _{fin} + R _{fout} 0.080 13 to 27 0.060 6.9 to 19 0.040 4.2 to 6.9	3.0 to 5.0	8.8 to 26	8.8 to 26	6.0 to 10.1 ^c	d.	6.6

a. Cylinder end must have more than one suction or discharge pocket.

b. If breather is not used, A_{port} = Valve Pocket Area.

c. If breather is not used, range of (R_{fin} + R_{fout}) is 8.8 to 26.

d. Value not available. Use A_{port} and (R_{fin} + R_{fout}) for Fig. 2. Result will be conservatively high.

$$U = 1.87 \times 10^{-24} (R_f \text{ in} + R_f \text{ out}) \frac{PM}{ZT} \times \frac{(A_{\text{pist.}} \cdot S_{\text{rpm}})^3}{(n \cdot A_{\text{port}})^2} \quad (17)$$

(constant = 3.30×10^{-16})

By the conditions of derivation, equation (17) is perfectly general. Table 1 includes representative values of resistance factors, R_f, for various mechanical configurations for bypass deactivation.

Resistance factors may be determined in the field, by recording the pressure cycle in a deactivated cylinder end during dynamic operation. Once a pressure-stroke diagram is available, ΔP_{pred} mean for the in and out strokes may be determined and resistance factors calculated by equation (14). Passage pressure should also be recorded so that pulsation effects present may be isolated

and deducted. See the following section for a discussion of the possible influence of pulsation.

INFLUENCE OF PULSATION

As previously stated, the derivation of power requirement is based on the assumption that a constant pressure prevails in the gas passage, suction or discharge, into which the bypass is directed. Departure from this condition can result in an additional power requirement, the magnitude of which is dependent upon the amplitude and phasing of the wave. Figs. 10(a) and 10(b) illustrate the point. In view of the foregoing, an objective of gas piping design should be to minimize pulsation influence in the bypass passage during deactivation. Modern piping system design practices, such as those involving the Southern Gas Association electro-acoustical compressor system analog and associated technology, afford

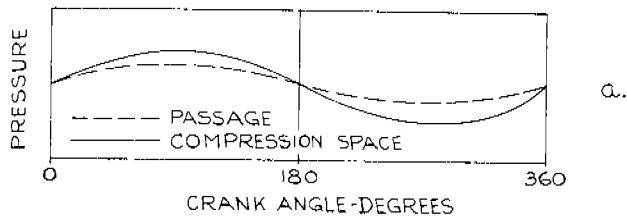


Fig. 10(a) Illustrative pressure-time traces of bypass system with gas pulsation present

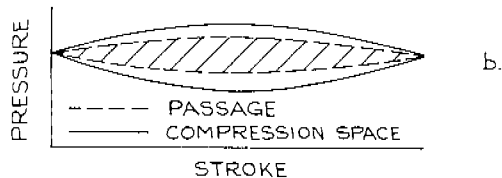


Fig. 10(b) Pressure-stroke replot of Fig. 10(a). Cross-hatched area represents work requirement due to presence of pulsation. Uncross-hatched areas represent work requirement due to pressure drops across bypass port(s)

the means of accomplishing this. --

Fig. 11 is a pressure-stroke diagram taken from a cylinder and deactivated by finger-type suction valve unloaders in a field test. A comparison of power requirements from the field test and those calculated by equation (17) using resistance factors determined by steady flow laboratory determinations is shown in Table 2 along with cylinder particulars and operating conditions.

EFFECT OF DEACTIVATED CYLINDER END ON PERFORMANCE OF ACTIVE ENDS OPERATING IN PARALLEL WITH IT

Power consumed by a deactivated cylinder end manifests itself in heating gas involved in the bypass operation. If, as an upper limit, it is assumed that none of this heat is lost to jacket coolant or the surroundings, gas flow of active ends operating parallel will be heated by an amount,

$$\Delta T = \frac{U \times 10^3}{N C_p} \quad (18)$$

(constant = 42.4)

This influence is obviously most direct in the case of a double-acting cylinder with one end deactivated. When the bypass is to discharge, a maximum increase in discharge temperature in the amount, ΔT , could be experienced.

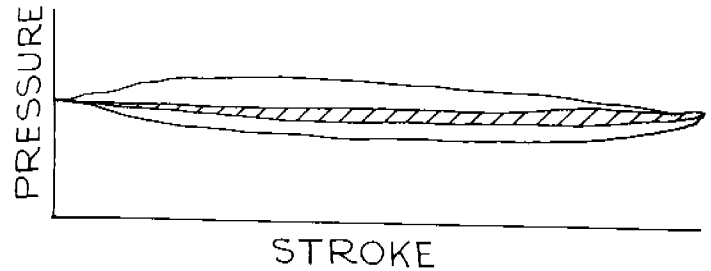


Fig. 11 Field test pressure-stroke diagram of cylinder end deactivated by finger-type suction valve unloaders

In the case of bypass to suction, suction temperature of the active end could also be increased by ΔT as a maximum. Assuming compression in the active end to occur isentropically, discharge temperature would then be

$$T_D = (T_s + \Delta T) \left(\frac{P_D}{P_s} \right)^{\frac{k-1}{k}} \quad (19)$$

Further, in the case of bypass to suction, heating of the inlet gas to the active end reduces its capacity to a fraction, F , of that obtained with no bypass preheating.

$$F = \frac{T_s}{T_s + \Delta T} \quad (20)$$

Table 2 Comparison of Field Test and Calculated Performance

Event	Power Requirement -- Kw (Hp)*			
	Field Test		Calculated	
In	19	(26)	17	(23)
Out	26	(34)	26	(35)
Pulsation	13	(17)	--	--
Total	58	(77)	43	(58)

* Cylinder and Application Details. Diameter = 584 mm (23 in.); stroke = 508 mm (20 in.); cylinder end -- head; rpm = 250; absolute suction pressure = 883 kilopascals (128 psi); molecular weight = 19.9; suction temperature = 283 K (510 R).

ADDITIONAL CONSIDERATIONS

Gas Temperature in the Deactivated End

Equations (11), (12), (14), and (17) relate to pressure drop or power requirement during dynamic operation and all include absolute temperature, T , in the denominator. Unless actual temperature in the deactivated end is known or can be esti-

mated, it is suggested that nominal suction or discharge gas temperatures be used in these equations. This will result in predicting conservatively high values for pressure drop and power.

crosshead pin loading. particularly with respect to non-reversing pin loading, should be investigated prior to a final decision.

With regard to bypass to discharge, it must be recognized that internals of the unloaded end are exposed to a temperature in excess of normal discharge temperature. Care must be taken not to exceed the temperature that can be tolerated by the lubricant or compressor components, particularly non-metallics.

Deactivation of an Entire Cylinder

Circulated jacket coolant is required where a single-acting cylinder or both ends of a double-acting cylinder are deactivated (brief deactivation during start-up excepted). Moreover, the coolant must be capable of removing the heat equivalent of work losses generated while maintaining cylinder internals at a safe temperature level. Consult the compressor manufacturer for guidance.

In connection with complete cylinder deactivation, the finger-type suction valve unloader, Fig. 2, is commonly used where "loadless starting" is required.

Examination of equation (17) reveals that power requirement is directly proportional to bypass pressure. Thus, it is advantageous to lower the pressure level in the gas system of the deactivated cylinder if possible.

Optimum Number of Bypass Devices per Cylinder

In cases where the bypass device is installed in a valve pocket and the end to be deactivated has more than one suction discharge valve pocket, a decision must be made regarding the number of bypass assemblies to use. In this connection, it is important to recall from equation (17) that the power requirement is inversely proportional to n^2 , where n is number of bypasses employed.

If the bypass configuration contains active valve area, careful consideration of this point is necessary to obtain the optimum balance between valve loss during active operation and power required during deactivation.

Which End of A Double-Acting Cylinder to Deactivate?

The required magnitude of power or capacity reduction may indicate the desirability of deactivating head or crank end. However, the effects on piston rod and

NOMENCLATURE

A_{pist} = piston area, sq m (sq in.)	R_f = resistance factor, dimensionless
A_{port} = bypass port area of one device, sq mm (sq in.)	R_{fin} = resistance factor for flow into compression space
C_p = specific heat capacity at constant pressure, joules/kg deg K (British thermal units per pound deg R)	R_{fout} = resistance factor for flow out of compression space
F = fraction of normal capacity obtained with bypass heating present	S = stroke, millimeters (in.)
L = work per stroke, joules (ft-lb)	T = absolute temperature, deg K (deg R)
M = molecular weight	T = temperature change, deg K (deg R)
N = gas flow rate, kg/sec (ppm)	U = power, kw (hp)
P = absolute pressure, kilopascals (psi)	$V_{pist.}$ = instantaneous piston velocity, m/sec (fpm)
ΔP = pressure change, kilopascals (psi)	κ = compressibility factor, dimensionless
ΔP_{ideal} = ideal pressure drop	g = acceleration ₂ constant, 9.81 m/sec ² (32.2 ft/sec ²)
$\Delta P_{obs.}$ = observed pressure drop	h = head, m (ft)
$\Delta P_{pred.}$ = predicted pressure drop corresponding to a given instantaneous piston velocity	k = gas property as defined by $(\frac{T_D}{T_S}) = (\frac{P_D}{P_S})^{\frac{k-1}{k}}$ over an isentropic path, dimensionless
$\Delta P_{pred. mean}$ = mean pressure drop over entire stroke of pressure-stroke diagram	n = number of bypass ports, dimensionless
Q = gas flow rate, cu m/sec (cfm)	rpm = rotative speed, rpm
R = gas constant, 8.31 x 10 ³ joules per kg mole deg K (1545 British thermal units per pound mole deg R)	ρ = density, kg/cu m (lb/cu ft)

Subscripts

D = discharge condition
S = suction condition